I, Akio Miyazawa, residing at 4-1-20-505, Zushi, Zushi-shi, Kanagawa 249-0006, Japan, and working for ISP Corporation of 1-29, Akashi-cho, Chuo-ku, Tokyo 104-0044, Japan, fully conversant with the English and Japanese languages, do hereby certify that to the best of my knowledge and belief the following is a true translation of Japanese Patent Application No. 2002-207345 filed in the Japanese Patent Office on the 16th day of July, 2002 in respect of an application for Letters Patent.

Signed, this 19th day of December, 2005

Akio Miyazawa

[Document Name]

[Title of Invention]

Gear Transmission for Automatic Transmission

[Scope of Claims]

[Claim 1] In a gear transmission for an automatic transmission which comprises an input portion receiving rotation from a power source, an output portion coaxially arranged with the input portion and transferring output rotation of the gear transmission, a plurality of planetary gearsets which are capable of providing a plurality of transmission paths between the input portion and the output portion and have a compound planetary gear train, and three clutches and two brakes which are capable of being selectively engaged or disengaged so that the rotation from the input portion is changed at a corresponding gear ratio and is output to the output portion through the selection of one of the transmission paths by the plurality of planetary gearsets, and which is arranged to select at least forward six gear and a reverse gear by the combination of the engagement/disengagement of these brakes and clutches,

the gear transmission of the automatic transmission being characterized in that

a reduction planetary gearset for reducing and outputting speed of rotation input is constructed by one of the plurality of the planetary gearsets,

a double-sun-gear type planetary gearset is constructed by one of the planetary gearsets in the compound planetary gearset constructed by tow sets of the planetary gearset comprises two sun gears, common pinions which mesh with the two sun gears, one ring gear which meshes with the pinions, and a carrier which rotatably supports the pinions,

a single-pinion type planetary gear set of the other planetary gearset is constructed by one sun gear, pinions meshing with the sun gear, and a carrier rotatably supporting the pinions,

a ring gear is an input member of the reduced rotation from the reduction planetary gearset to the compound planetary gear train,

two clutches, of the three clutches, for directing and cutting the reduced rotation are disposed at a radially outside portion of the compound planetary gear train, and

the two brakes are disposed at a radially outside portion of the two clutches of directing and cutting the reduced rotation, such that one of the two brakes and one of the two brakes overlap at least partially in the axial direction and the other of the two clutches and the other of the two brakes overlap at least partially in the axial direction.

[Claim 2] The gear transmission for the automatic transmission claimed in claim 1, characterized in that the two brakes are disposed at radially outside portions of the two clutches for directing and cutting the reduced rotation, such that a friction member of one of the clutches and a friction member of one of the brakes overlap at least partially in the axial direction and a friction member of the other of the clutches and a friction member of the other of the brakes overlap at least partially in the axial direction.

[Claim 3] The gear transmission for the automatic transmission claimed in claim 1 or claim 2, characterized in that the output member of the compound planetary gear train is disposed at a radially outside portion of the two clutch

for directing and cutting the reduced rotation and a radially inside portion of the two brakes.

[Claim 4] The gear transmission for the automatic transmission claimed in claim 3, being characterized in that the output member is an output drum of the double-sun-gear type planetary gearset and the single-pinion type planetary gearset.

[Detailed Explanation of the Invention]

[0001]

[Technical Field of the Invention]

The present invention relates to techniques for improving a compactness of a gear transmission for an automatic transmission which is constructed by an input portion, a plurality of planetary gearsets, three clutches, two brakes and an outputs portion, and which is capable of producing at least six forward speeds and one reverse speed by properly engaging and disengaging the three clutches and the two brakes of shift elements.

[0002]

[Prior Art]

Recently, there is a tendency that an automatic transmission is constructed to have more stepped speeds for improvement in fuel efficiency and drivability and fuel. For example, Japanese Patent Provisional Publication 2000-55152 shown in Fig. 9 discloses an automatic transmission where six forward speeds and reverse are selectable and employs a Ravigneaux compound planetary gear train (compound planetary train in which two sets of planet-pinions are meshed respectively with different sun gears. The automatic transmission disclosed therein comprises clutches C1 and C2 which are selectably engageable to direct and cut rotation

that has been reduced in speed. The clutches C1 and C2 are disposed behind and around the planetary gear train G. A brake B1 (band type brake in Figure) is disposed about an outer circumference of the Ravigneaux type compound gear train G and a brake B2 is disposed in a single row similarly.

[0003]

[Problem to be solved by the Invention]

However, according to the related art, the clutches C1 and C2, and the brake B2 (clutch type brake) are disposed in a row in the axial direction around an outer periphery of the planetary gear train G, and this leads to a problem of the axial length of the transmission being large. Also, one of the planetary gearsets constituting the planetary gear train G is a double-pinion planetary gearset, and rotation having an increased torque after being reduced in speed by the reduction planetary gearset is input to the planetary gear train G from sun gears S2 and S3 thereof, meaning that an outer diameter of the planetary gear train G must be large, and consequently the transmission becomes undesirably large overall.

[0004]

It is therefore an object of the present invention to effectively solve the above problems and to improve a compactness of a gear transmission for an automatic transmission which is constructed by an input portion, a plurality of planetary gearsets, three clutches, two brakes and an outputs portion, and which is capable of producing at least six forward speeds and one reverse speed by properly engaging and disengaging the three clutches and the two brakes of shift elements.

[0005]

[Means for solving the Problem]

For this purpose, the gear transmission for the automatic transmission according to the present invention, as claimed in claim 1, comprises an input portion receiving rotation from a power source, an output portion coaxially arranged with the input portion and transferring output rotation of the gear transmission, a plurality of planetary gearsets which are capable of providing a plurality of transmission paths between the input portion and the output portion and have a compound planetary gear train, and three clutches and two brakes which are capable of being selectively engaged or disengaged so that the rotation from the input portion is changed at a corresponding gear ratio and is output to the output portion through the selection of one of the transmission paths by the plurality of planetary gearsets, and is arranged to select at least six forward gear and a reverse gear by the combination of the engagement/disengagement of these brakes and clutches, as presumption.

[0006]

In the present invention, a reduction planetary gearset for reducing and outputing speed of rotation input is constructed by one of the plurality of the planetary gearsets.

A double-sun-gear type planetary gearset is constructed by one of the planetary gearsets in the compound planetary gearset constructed by tow sets of the planetary gearset comprises two sun gears, common pinions which mesh with the two sun gears, one ring gear which meshes with the pinions, and a carrier which rotatably supports the pinions,

A single-pinion type planetary gearset of the other planetary gearset is constructed by one sun gear, pinions meshing with the sun gear, and a carrier rotatably supporting the pinion.

A ring gear is an input member of the reduced rotation from the reduction planetary gearset to the compound planetary gear train.

Further, in the present invention, two clutches for directing and cutting the reduced rotation, of the three clutches, are disposed at a radially outside portion of the compound planetary gear train, and

the two brakes are disposed at a radially outside portion of the two clutches of directing and cutting the reduced rotation, such that one of the two brakes and one of the two brakes overlap at least partially in the axial direction and the other of the two clutches and the other of the two brakes overlap at least partially in the axial direction.

[0007]

[Advantage of the Invention]

With the gear transmission of the present invention, since the two brakes are disposed at radially outside portions of the two clutches of directing and cutting the reduced rotation, such that one of the two brakes and one of the two brakes overlap at least partially in the axial direction and the other of the two clutches and the other of the two brakes overlap at least partially in the axial direction, it becomes possible to shorten the dimension of the transmission in the axial direction. Further, since it is possible to bring the two brakes close to the two clutches of directing and cutting the reduced rotation in

the axial direction, layout of fluid passages is facilitated in the axial direction and fluid passage structure is thereby simplified.

[0008]

With respect to the radial dimension of the transmission, since the planetary gearsets G2 and G3 which constitute the compound planetary gear train are single-pinion type, the compound planetary gear train can be designed with a smaller diameter. Moreover, by making the third planetary gearset G3 of the compound planetary gearset a double-sun-gear type planetary gearset, it is possible for the second ring gear R2 to serve as the input member of reduced rotation from the reduction planetary gearset G1 to the compound planetary gearsets G2 and G3. Compared to a sun gear serving as an input member, there is less tangential stress present with a ring gear acting as the input member, and is therefore advantageous with respect to a number of points including gear strength, gear life, and carrier rigidity, and it is possible to make a diameter of the compound planetary gear train smaller.

[0009]

It is preferable that the gear transmission for the automatic transmission has a construction that the two brakes are disposed at radially outside portions of the two clutches for directing and cutting the reduced rotation, such that a friction member of one of the clutches and a friction member of one of the brakes overlap at least partially in the axial direction and a friction member of the other of the clutches and a friction member of the other of the brakes overlap at least partially in the axial direction, as claimed in claim 2.

[0010]

The gear transmission for the automatic transmission may be constructed such that the output member of the compound planetary gear train is disposed at radially outside portions of the two clutch for directing and cutting the reduced rotation and at radially inside portions of the two brakes, as claimed in claim 3. This allows the output member to be made with a larger diameter, which is an advantage with respect to strength. With a larger diameter, it can then be designed with a smaller thickness and still retain sufficient strength for transmitting high torque. This makes it possible to more effectively design a smaller transmission.

[0011]

The gear transmission for the automatic transmission may be constructed such that the output member is an output drum of the double-sun-gear type planetary gearset and the single-pinion type planetary gearset, as claimed in claim 4.

[0012]

[Embodiment of the Invention]

Hereinafter, there is explained an embodiment of the present invention on the basis of the drawings, in detail.

Fig. 1 schematically shows a gear transmission for an automatic transmission which is an embodiment of the present invention. G1 denotes a first planetary gearset, G2 denotes a second planetary gearset, G3 denotes a third planetary gearset, M1 denotes a first connecting member, M2 denotes a second connecting member, C1 denotes a first clutch, C2 denotes a second clutch, C3 denotes a third clutch, B1 denotes a first brake, B2 denotes a second brake, Input denotes an input member Input (an input shaft 1), and Output

denotes an output member (an output gear 2). In this embodiment, the gear transmission for the automatic transmission is constructed as a gear transmission for an automatic transmission of a vehicle.

[0013]

The gear transmission for an automatic transmission according to the present embodiment comprises, starting from the left end portion (end portion near the input member Input) of Fig. 1, the first planetary gearset G1 as a reduction mechanism comprised of a single-pinion planetary gearset having a single set of planet-pinions, the second planetary gearset G2 comprised of a single-pinion planetary gearset also having a single set of planet-pinions, and the third planetary gearset G3 comprised of a double-sun-gear type planetary gearset having two sun gears, all being disposed coaxially. The first planetary gearset G1 functions as a reduction planetary gearset, and the second planetary gearset G2 and the third planetary gearset G3 constitute a compound planetary gear train located in a rear portion of the transmission.

[0014]

The first planetary gearset G1 is a single-pinion planetary gearset (a reduction planetary gearset) comprised of a first sun gear S1, a first ring gear R1, the first planet-pinions P1 which mesh with the first sun gear S1 and the first ring gear R1, and a first carrier PC1 which supports the first planet-pinions P1 to be freely rotatable. The second planetary gearset G2 is a single-pinion planetary gearset comprised of a second sun gear S2, a second ring gear R2, the second planet-pinions P2 which mesh with the second sun gear S2 and the second ring gear R2, and a second

carrier PC2 which supports second planet-pinions P2 to be freely rotatable.

[0015]

The third planetary gearset G3 is a double-sun-gear planetary gearset comprised of a third sun gear S3 which is located at a near side of the input member Input and a fourth sun gear S4 which is located at a far side of the input member Input, the third planet-pinions P3 which mesh both with the third sun gear S3 and the fourth sun gear S4, a third carrier PC3 which supports the third planet-pinions P3 to be freely rotatable, and a third ring gear R3 which meshes with the third planet-pinions P3.

The third sun gear S3 and the fourth sun gear S4 are disposed coaxially, but it is not necessary for the third sun gear S3 and the fourth sun gear S4 to have the same number of teeth.

There are provided a center member CM which extends radially inward toward the axis from between the third sun gear S3 and the fourth the sun gear S4 and an outer member OM which extends radially outward away from the axis. The outer member OM is uniquely disposed, and this will be discussed in more detail.

Further, the center member CM extends radially inward toward the axis so that it passes through space existing between individual planet-pinions of the third planet-pinions P3.

[0016]

The input member Input comprises the input shaft 1, and the input shaft 1 is joined to the first ring gear R1, and is coupled to a not-shown engine through a torque converter

(not shown), so that engine rotation is input into the first ring gear R1 from the input shaft 1.

The output member Output comprises the output gear 2, and is joined coaxially to the second connecting member M2 which serves to join the second carrier PC2 and the third ring gear R3. The output rotation from the transmission is transmitted, for example, from the output gear 2 to the counter gear 30 shown in Fig. 6, then on to a final gear and differential gear apparatus (neither of which shown) to the drive wheels of an automobile.

Further, the first connecting member M1 serves as a connecting member to join the second sun gear S2 and the third sun gear S3 to form a single integral body.

[0017]

The first sun gear S1 of the reduction planetary gearset G1 is permanently fixed to the transmission case 3, and the first carrier PC1 is appropriately joinable to the second ring gear R2 by the first clutch C1 and is appropriately joinable to the second sun gear S2 by the second clutch C2.

The center member CM of the third carrier PC3 is appropriately joinable to the input shaft 1 by the third clutch C3, and therefore the third clutch C3 serves as a direct clutch to transmit input rotation directly to the compound planetary gear train comprised of the second planetary gearset G2 and the third planetary gearset G3.

The outer member OM of the third carrier PC3 of the third planetary gearset G3, which is a double-sun-gear planetary gearset, is appropriately joinable to the transmission case 3 by the first brake B1 so that the third carrier PC3 is made appropriately fixable, and the fourth

sun gear S4 is made appropriately fixable to the transmission case 3 by the second brake B2.

[0018]

It is possible to select gears (forward speeds 1st through 6th and reverse) with the gear transmission through the corresponding combinations of the clutches C1, C2 and C3, and the brakes B1 and B2, as shown by the engagement logic table in Fig. 2, where engagement is represented by a circle mark and disengagement by being unmarked. A control valve body (not shown) for controlling gear shift is connected to the clutches and the brakes to realize the engagement logic.

A hydraulic type, an electronic type, or a combination type which combines these two types are employable as a control valve body for control of gear shift.

[0019]

Hereinafter, operation of the gear transmission according to the present invention will be explained with reference to Figs. 2 through 5.

Fig. 2 shows the engagement logic table of shift element in the gear transmission. Figs. 3 through 5 are explanatory views showing a torque transmission path at each gear of the gear transmission.

In Figs. 3 through 5, the torque transmission path of each of the clutches, the brakes and the members is indicated by thick lines, and gears which participate in torque transmission are indicated by hatching.

[0020]

(First gear)

As shown by Fig. 2, forward first gear is achieved through engagement of the first clutch C1 and the first brake B1.

In first gear, reduced rotation from the first planetary gearset G1 is input into the second ring gear R2 of the second planetary gearset G2 by engagement of the first clutch C1.

At the same time, the third carrier PC3 of the third planetary gearset G3 is fixed to the transmission case 3 by engagement of the first brake B1, so the rotation of the third sun gear S3 becomes a reverse-direction reduced rotation relative to the output rotation of the third ring gear R3. The rotation of the third sun gear S3 is then transmitted to the second sun gear S2 of the second planetary gearset G2 via the first connecting member M1.

Thus, at the second planetary gearset G2, a normal-direction reduced rotation is input from the second ring gear R2, and a reverse-direction reduced rotation is input from the second sun gear S2, and as a result, a rotation which is a further reduced rotation from the second ring gear R2 is output to the output gear 2 via the second connecting member M2 from the second carrier PC2.

[0021]

The torque transmission path in first gear is as shown in Fig. 3(a). Torque acts through the first clutch C1, the first brake B1, the first connecting member M1, and the second connecting member M2, shown in bold lines, and the first planetary gearset G1, the second planetary gearset G2, and the third planetary gearset G3 not including the fourth sun gear S4, shown in hatching.

Therefore, in first gear, all planetary gearsets are involved in transmission of torque, that is, the first planetary gearset G1, as well as the second planetary gearset G2 and the third planetary gearset G3 which make up

the compound planetary gear train located in the rear portion of the transmission.

[0022]

(Second gear)

As shown by Fig. 2, in second gear, the first brake B1 which was engaged in first gear is disengaged, and the second brake B2 is engaged instead. Therefore, second gear is achieved through engagement of the first clutch C1 and the second brake B2.

In second gear, reduced rotation from the first planetary gearset G1 is input into the second ring gear R2 of the second planetary gearset G2 by engagement of the first clutch C1.

At the same time, the fourth sun gear S4 of the third planetary gearset G3 is fixed to the transmission case 3 by engagement of the second brake B2, so the third sun gear S3 connected to the fourth sun gear S4 by the third planet-pinions P3 is fixed. The second sun gear S2 which is joined to the third sun gear S3 by the first connecting member M1 is then fixed to the transmission case 3.

Thus, at the second planetary gearset G2, normal-direction reduced rotation is input from the second ring gear R2, and the second sun gear S2 is fixed, and as a result, a reduced rotation from the second ring gear R2 which has been further reduced (this rotation is faster than the rotation in first gear) is output to the output gear 2 through the second connecting member from the second carrier PC2.

[0023]

The torque transmission path in second gear is as shown in Fig. 3(b). Torque acts through the first clutch C1, the

second brake B2, the first connecting member M1, and the second connecting member M2, shown in bold lines, and the first planetary gearset G1 and the second planetary gearset G2, shown in hatching.

Further, regarding the third planetary gearset G3, the third planet-pinions P3, which are not constrained, are made to revolve accompanying output rotation of the third ring gear R3, and thus revolve about the third sun gear S3 and the fourth sun gear S4 which are both fixed. Also, torque which constrains the second sun gear S2 acts through M1-S3-P3-S4.

[0024]

(Third gear)

In third gear, as shown in Fig. 2, the second brake B2 which was engaged in second gear is disengaged, and the second clutch C2 is engaged instead. Therefore, third gear is achieved through engagement of the first clutch C1 and the second clutch C2.

In third gear, reduced rotation from the first planetary gearset G1 is input into the second ring gear R2 of the second planetary gearset G2 by engagement of the first clutch C1. Simultaneously, by engagement of the second clutch C2, this reduced rotation is input into the second sun gear S2 of the second planetary gearset G2.

Thus, at the second planetary gearset G2, by the same reduced rotation being input from the second ring gear R2 and the second sun gear S2, the second carrier PC2 rotates integrally therewith, and reduced rotation (which is the same as the reduced rotation from planetary gearset G1) is input to the output gear 2 through the second connecting member M2.

The torque transmission path in third gear is as shown in Fig. 3(c). Torque acts through the first clutch C1, the second clutch C2, and the second connecting member M2, shown in bold lines, and the first planetary gearset G1 and the second planetary gearset G2, shown in hatching. The third planetary gearset G3 does not participate in transmission of torque.

[0025]

(Fourth gear)

In fourth gear, as shown in Fig. 2, the second clutch C2 which was engaged in third gear is disengaged, and the third clutch C3 is engaged instead. Therefore, fourth gear is achieved by engagement of the first clutch C1 and the third clutch C3.

In fourth gear, reduced rotation from the first planetary gearset G1 is input into the second ring gear R2 of the second planetary gearset G2 by engagement of the first clutch C1.

At the same time, input rotation from the input shaft 1 is input into the third carrier PC3 of the third planetary gearset G3 through the center member CM by engagement of the third clutch C3. As a result, rotation of the third sun gear S3 is faster than output rotation of the third ring gear R3, and this faster rotation of the third sun gear S3 is transmitted to the second sun gear S2 through the first connecting member M1.

[0026]

Thus, at the second planetary gearset G2, reduced rotation from the second ring gear R2 is input, and faster rotation is input from the second sun gear S2, and as a result, rotation which is a faster reduced rotation (which

is slower than the input rotation from input shaft 1) is from the second ring gear R2 is output to the output gear 2 from the second carrier PC2 through the second connecting member M2.

The torque transmission path in fourth gear is as shown in Fig. 4(a). Torque acts through the first clutch C1, the third clutch C3, the center member CM, the first member M1, and the second member M2, shown in bold lines, and the first planetary gearset G1, the second planetary gearset G2, and the third planetary gearset G3 (not including the fourth sun gear S4), shown in hatching.

[0027]

(Fifth gear)

In fifth gear, as shown in Fig. 2, the first clutch C1 which was engaged in fourth gear is disengaged, and the second clutch C2 is engaged instead. Therefore, fifth gear is achieved by engagement of the second clutch C2 and the third clutch C3.

In fifth gear, reduced rotation from the first planetary gearset G1 is input into the third sun gear S3 through the second sun gear S2 and the first connecting member M1 by engagement of the second clutch C2. Simultaneously, input rotation from the input shaft 1 is input into the third carrier PC3 through the center member CM by engagement of the third clutch C3.

[0028]

Thus, at the third planetary gearset G3, input rotation is input into the third carrier PC3, and reduced rotation is input into the third sun gear S3 from the first planetary gearset G1, and as a result, rotation which is faster than

the input rotation is output to the output gear 2 from the third ring gear R3.

The torque transmission path in fifth gear is as shown in Fig. 4(b). Torque acts through the second clutch C2, the third clutch C3, the center member CM, and the first connecting member M1, shown in bold lines, and the first planetary gearset G1, the second sun gear S2, and the third planetary gearset G3 (not including the fourth sun gear S4), shown in hatching.

[0029]

(Sixth gear)

In sixth gear, as shown in Fig. 2, the second clutch C2 which was engaged in fifth gear is disengaged, and the second brake B2 is engaged instead. Therefore, sixth gear is achieved by engagement of the third clutch C3 and the second brake B2.

In sixth gear, input rotation from the input shaft 1 is input into the third carrier PC3 through the center member CM of the third planetary gearset G3 by engagement of the third clutch C3. Also, the fourth sun gear S4 of the third planetary gearset G3 is fixed to the transmission case 3 by engagement of the second brake B2.

[0030]

Thus, at the third planetary gearset G3, input rotation is input into the third carrier PC3, and the fourth sun gear S4 is fixed to the transmission case 3, and as a result, rotation which is faster than the input rotation is output to the output gear 2 from the third ring gear R3.

The torque transmission path in sixth gear is as shown in Fig. 4(c). Torque acts through the third clutch C3, the second brake B2, and the center member CM, shown in bold

lines, and the third planetary gearset G3 (not including the third sun gear S3), shown in hatching.

[0031]

(Reverse gear)

As shown in Fig. 2, reverse gear is achieved by engagement of the second clutch C2 and the first brake B1.

In reverse gear, reduced rotation from the first planetary gearset G1 is input into the third sun gear S3 through the second sun gear S2 and the first connection member M1 by engagement of the second clutch C2. At the same time, by engagement of the first brake B1, the third carrier PC3 is fixed to the transmission case 3.

Thus, at the third planetary gearset G3, normal-direction reduced rotation is input into the third sun gear S3, and the third carrier PC3 is fixed to the transmission case 3, and as a result, reverse-direction rotation that has been reduced in speed is output to the output gear 2 from the third ring gear R3.

[0032]

The torque transmission path in reverse gear is as shown in Fig. 5. Torque acts through the second clutch C2, the first brake B1, the first connecting member M1, and the outer member OM, shown in bold lines, and the first planetary gearset G1, the second sun gear S2, and the third planetary gearset G3 (except the fourth sun gear S4), shown in hatching.

[0033]

Fig. 6 is an embodiment structural view of the gear transmission. Fig. 7 is a schematic skeleton diagram of the gear transmission for an automatic transmission shown in Fig. 1 and shows an essential part of the embodiment according to

the present invention. Fig. 8 is an enlarged view showing a part relating the present invention in the construction of the embodiment.

Hereinafter, there is discussed the embodiment construction of the gear transmission on the basis of there figures. In Figs. 6 and 8, the gear transmission is shown such that the orientation of the input and output portions is opposite to that of the skeleton diagrams of Figs. 1, 3 through 5, and 7.

The input shaft 1 and a middle shaft 4 are disposed in transmission case 3 so that a rear end of input shaft 1 is supported in a front end of the middle shaft 4 to form a fitting portion, such that the input shaft 1 is coaxially rotatable relative to the middle shaft 4. The input shaft 1 and the middle shaft 4 are supported to be individually and rotatable with respect to the transmission case 3.

[0034]

A front end opening of the transmission case 3 near the input shaft 1 is closed by an oil pump which comprises a pump housing 5 and a pump cover 6, and the input shaft 1 is passed through the oil pump to be axially supported thereby. An end of the input shaft 1 protruding from the oil pump is connected to an engine (not shown) via a torque converter (not shown).

[0035]

A rear end of the middle shaft 4 which is away from the input shaft 1 is supported to be rotatable by an end cover 7 at a rear end of the transmission case 3.

A midway wall 8 is disposed approximately halfway axially inside the transmission case 3, and the output gear 2 is supported thereon to be rotatable. A hollow shaft 9 is

disposed in a center opening of the midway wall 8, and the butt fitting portion of the input shaft 1 and the middle shaft 4 is supported to be rotatable inside the hollow shaft 9 by the center hole of the midway wall 8.

[0036]

As shown in Figs. 6 and 7, the first planetary gearset G1 is disposed in a space existing in a front portion (a front end portion of the automatic transmission) between the oil pump comprised of the pump housing 5 and the pump cover 6 and the midway wall 8, and the third clutch C3 is disposed so as to enclose the first planetary gearset G1.

Referring to the first planetary gearset G1, the first sun gear S1 is serration fitted to a center boss portion 6a projecting from a rear of the pump cover 6 to be permanently non-rotatable so as to function as a reaction force stopper, and the first ring gear R1 which is a rotation input member is joined to an outer perimeter of a flange 10 which extends radially outward away from the axis from the input shaft 1.

[0037]

A clutch drum 11 extends radially outward away from the axis from a front end of the middle shaft 4 near the input shaft 1 and encloses the first ring gear R1. The third clutch C3, which serves as a direct clutch, is disposed about an outer circumference of the first planetary gearset G1, the third clutch C3 comprising a clutch pack 12 and a clutch piston 13 which will be discussed later. The clutch pack 12 comprises alternating clutch plates respectively splined to an inner circumference of the clutch drum 11 and an outer circumference of the first ring gear R1.

The first ring gear R1 also serves as a clutch hub of the third clutch C3.

Further, the clutch piston 13, which is a clutch piston of the third clutch C3, is disposed on a side of the first planetary gearset G1 away from the oil pump which is comprised of the pump housing 5 and the pump cover 6, and the clutch piston 13 is slidably fitted to a front end of the middle shaft 4 and a cylinder 11a of the clutch drum 11 which faces the first planetary gearset G1.

[0038]

The third clutch C3 is engageable by the third clutch piston 13 traveling to the right of Fig. 6 after having received line pressure supplied via a fluid passage 14 from a control valve body.

A connecting shell 53, which is drum-shaped, extends radially outward away from the axis from a front end of the hollow shaft 9, continues on to enclose the third clutch C3, and continues further so that a front end of the connecting shell 53 is joined to the first carrier PC1.

The first carrier PC1, as obvious from the previous explanation, constitutes a rotation output member of the first planetary gearset G1 (a reduction planetary gearset).

[0039]

As shown in Figs. 6 through 8, the first clutch C1, the second clutch C2, the first brake B1, and the second brake B2 are disposed in a space which exists between the midway wall 8 and the end cover 7.

[0040]

The second planetary gearset G2 and the third planetary gearset G3 are disposed about the middle shaft 4, the second planetary gearset G2 being positioned nearer to the input shaft 1 than the third planetary gearset G3.

The second sun gear S2 of the second planetary gearset G2 and the third sun gear S3 of the third planetary gearset G3 are joined to form a single integral body by the first connecting member M1 and are supported to be rotatable by the middle shaft 4.

A clutch drum 15 extends radially outward away from the axis from approximately halfway of the hollow shaft 9, continues on to extend axially toward the rear of the transmission case 3, and continues further somewhat past an outer circumference of the second ring gear R2. The first clutch C1 is comprised of a clutch pack 16 and a clutch piston 19. The clutch pack 16 is comprised of alternating clutch plates respectively splined to an inner circumference of the clutch drum 15 and an outer circumference of the second ring gear R2.

[0041]

As discussed above, the second clutch C2 is disposed nearer to the input shaft 1 than the first clutch C1 which is disposed on an outer circumference of the second planetary gearset G2, so a clutch hub 17 which extends radially outward away from the axis is fixedly installed to an outer edge of an input shaft of the second sun gear S2. The second clutch C2 is comprised of a clutch pack 18 and a clutch piston 20 which will be discussed hereinafter. The clutch pack 18 is comprised of alternating clutch plates respectively splined to an inner circumference of the clutch drum 15 and an outer circumference of the clutch hub 17.

Further, the clutch piston 19 of the first clutch C1 and the clutch piston 20 of second clutch C2 form a double piston which is disposed on a side of second clutch C2 away from first clutch C1, clutch piston 20 being slidable on an

inner side of clutch piston 19. The clutch piston 19 is fitted to be freely slidable on an end wall of clutch drum 15 which faces the second planetary gearset G2.

The first clutch C1 and the second clutch C2 are individually engageable by the clutch piston 19 and the clutch piston 20 traveling to the left of Fig. 6 after receiving line pressure from individual fluid passages of the plurality of fluid passages 21 (only one fluid passage is seen in Figure) which are formed in the midway wall 8 and the hollow shaft 9.

[0042]

As mentioned before, the third planetary gearset G3 is a double-sun-gear planetary gearset. However, the third ring gear R3 is fabricated so that a face width of the teeth thereof is smaller than a face width of the teeth of the third planet-pinions P3. By meshing the third ring gear R3 with the third planet-pinions P3 at an end portion of the third planet-pinions P3 near the second planetary gearset G2, the second connecting member M2 which joins the third ring gear R3 and the second carrier PC2 of the second planetary gearset G2 can be designed smaller.

A cylindrical connecting member (output drum) 22 is disposed so as to enclose the clutch drum 15 of the first clutch C1 and the second clutch C2, and serves as an output member of the compound planetary gear train which comprises the second planetary gearset G2 and the third planetary gearset G3. One end of the output drum 22 is attached to an outer circumference of the third ring gear R3, and another end thereof is attached to the output gear 2.

[0043]

The output member of the compound planetary gear train is disposed radially outside the respective outer circumferences of the two clutches of connecting/disconnecting the reduced rotation (first clutch C1 and second clutch C2) and radially inside the respective inner circumferences of two brakes (first brake B1 and second brake B2) which are discussed later in detail. This enables the output drum 22 to be formed with a large diameter, which is favorable in terms of strength. Thus, according to the embodiment of the present invention, a thickness of the output drum 22 is designed to be smaller than is generally so in the related art, while retaining sufficient strength characteristics.

[0044]

As has been discussed, the center member CM is disposed on the third carrier PC3 of the third planetary gearset G3 to extend radially inward toward the axis between the third sun gear S3 and the fourth sun gear S4, and the outer member OM is disposed on the third carrier PC3 at a position approximately halfway axially of the third planet-pinions P3 and extends radially outward away from the axis and along a rear face of the third ring gear R3.

The center member CM is connected to the middle shaft 4, and the third carrier PC3 is thereby connected to the clutch drum 11 of the third clutch C3 via the center member CM and the middle shaft 4.

A brake hub 23 is joined to an outer circumference of the outer member OM, and is disposed about an outer circumference of the cylindrical output drum 22 and extends toward the front of the transmission to within proximity of midway wall 8.

The first brake B1 is comprised of the brake pack 24 and the brake piston 25. The brake pack 24 is comprised of alternating plates respectively splined to an inner circumference of the brake hub 23 and an inner circumference of the transmission case 3. The brake piston 25 of the first brake B1 is fitted to the inside of the transmission case 3 behind the brake pack 24, and the first brake B1 is appropriately engageable by the brake piston 25.

[0045]

A brake hub 26 is disposed so as to overlap a rear end of the brake hub 23, and an end wall 26a of the brake hub 26 extends inward toward the axis along and behind the third planetary gearset G3, and an inner circumference of the rear wall 26a of the brake hub 26 is joined to the fourth sun gear S4 of the third planetary gearset G3.

The second brake B2 is comprised of a brake pack 27 and a brake piston 28. The brake pack 27 is comprised of alternating plates respectively splined to an inner circumference of the transmission case 3 and an outer circumference of the brake hub 26. The brake piston 28 of the second brake B2 is fitted to the inside of the transmission case 3 behind the brake pack 27, and the second brake B2 is appropriately engageable by the brake piston 28.

[0046]

Therefore, the first brake B1 and the second brake B2 are disposed around the second clutch C2 and around the first clutch C1 respectively, and the first brake B1 is disposed nearer to the input shaft 1 (first planetary gearset G1) than the second brake B2. The first brake B1 and the second brake B2 are disposed axially in a row and

nearer to the second planetary gearset G2 than the third planetary gearset G3.

[0047]

The disposition of the second planetary gearset G2, the third planetary gearset G3, the first clutch C1, the second clutch C2, the first brake B1, and the second brake B2, which are arranged as discussed above, will now be discussed in more detail.

As shown in Figs. 7 and 8, the two clutches (the first clutch C1 and the second clutch C2), which direct and cut reduced rotation, are disposed radially beyond the compound planetary gear train which is comprised of the second planetary gearset G2 and the third planetary gearset G3. These two clutches C1 and C2 are disposed outside and around the compound planetary gear train. The two brakes B1 and B2 are respectively disposed radially beyond the circumferences of first clutch C1 and second clutch C2, such that one of two clutches C1 and C2 and one of the brakes B1 and B2 overlap at least partially in the axial direction, and the other of the two clutches C1 and C2 and the other of the two brakes B1 and B2 overlap at least partially in the axial direction.

[0048]

More specifically, the second brake B2 is disposed around the circumference of the first clutch C1 so that the clutch pack 16 of the first clutch C1 and the brake pack 27 of the second brake B2 greatly overlap in the axial direction. Also, the second brake B2 is disposed far enough toward the rear of the transmission so that the clutch plate 18 of the second clutch C2 and the brake pack 24 of the first brake B1 overlap in the axial direction. Also, the

first clutch C1 and the second clutch C2 are positioned in a row axially.

[0049]

As shown in Fig. 8, working fluid to the first brake B1 is supplied through an opening member 51 which is disposed in the enclosing wall of the transmission case 3, and working fluid to the second brake B2 is supplied through an opening portion 52 which is disposed in the end cover 7. At the same time, lubricating oil flows radially outward away from the axis from within the middle shaft 4, and is supplied to the first clutch C1, the second clutch C2, the first brake B1, the second brake B2, and other elements. With that in consideration, the clutches C1 and C2, the brakes B1 an B2 are positioned axially close together to facilitate the layout of fluid passages with respect to the axial direction. The fluid passage structure is thus simplified, especially with regard to lubricating oil.

[0050]

Further, a one-way clutch OWC, which was omitted in the skeleton diagrams of Figs. 1 and 3 through 5, is disposed between the transmission case 3 and a front of the brake hub 23 which constitutes the first brake B1, as shown in Figs. 6, 7, and 8. Forward first gear is achievable even with the first brake B1 in a disengaged state due to the one-way clutch OWC stopping rotation of the third carrier PC3 in one direction.

However, while first gear can be achieved through provision of the one-way clutch OWC, the one-way clutch OWC allows reverse-direction rotation of the third carrier PC3 during engine braking. Engine braking is therefore not effective in this case. When engine braking is required,

the first brake B1 is engaged to stop reverse-direction rotation of the third carrier PC3.

A counter shaft 29 is supported to be rotatable inside the transmission case 3, and is parallel to the input shaft 1 and the middle shaft 4. A countergear 30 and a final drive pinion 31 are formed integrally with the counter shaft 29, the countergear 30 meshing with the output gear 2, and the final drive pinion 31 meshing with a differential gear assembly of drive wheels of a vehicle (not shown) are formed integrally with the counter shaft 29.

[0051]

According to the embodiment, referring to Figs. 6 through 8, at a position which is radially beyond and along the two clutches (the first clutch C1 and the second clutch C2) which direct and cut reduced rotation, one of the clutches C1 and C2 and one of the brakes B1 and B2 overlap at least partially in the axial direction, and the other of the clutches C1 and C2 and the other of the brakes B1 and B2 overlap at least partially in the axial direction. It therefore becomes possible to axially shorten the transmission case.

Since it is possible to axially approach the two brakes B1 and B2 and the two clutches C1 and C2 of directing and cutting the reduced rotation, layout of fluid passages is facilitated in the axial direction and fluid passage structure is thereby simplified.

[0052]

Further, the greater the overlap between one clutch and one brake (according to the embodiment, the first clutch C1 and the second brake B2) and between the other clutch and the other brake (according to the embodiment, the second

clutch C2 and the first brake B1), the more noticeable the beneficial effects of the present invention are. It is therefore preferable to dispose the clutches and the brakes to overlap as much as can be allowed by the particular arrangement of a given transmission assembly.

[0053]

With respect to the radial dimension of the transmission, the planetary gearsets G2 and G3 which constitute the compound planetary gear train are single-pinion type, so the compound planetary gear train can be designed with a smaller diameter. Moreover, by making the third planetary gearset G3 of the compound planetary gearset a double-sun-gear planetary gearset, it is possible for the second ring gear R2 to serve as the input member of reduced rotation from the reduction planetary gearset G1 to the compound planetary gearsets G2 and G3. Compared to a sun gear serving as an input member, there is less tangential stress present with a ring gear acting as the input member, and is therefore advantageous with respect to a number of points including gear strength, gear life, and carrier rigidity, and it is possible to make a diameter of the compound planetary gear train smaller.

[0054]

Also, in an automatic transmission according to the embodiment, the cylindrical connecting member (output drum) 22 acts as an output member of the compound planetary gear train which comprises the single-pinion planetary gearset G2 and the double-sun-gear planetary gearset G3, and is disposed radially beyond the respective outer circumferences of the two clutches (the first clutch C1 and the second clutch C2) as well as radially within the respective inner

circumferences of the two brakes. This allows the cylindrical connecting member 22 to be made with a larger diameter, which is an advantage with respect to strength. With a larger diameter, it can then be designed with a smaller thickness and still retain sufficient strength for transmitting high torque. This makes it possible to more effectively design a smaller transmission.

[0055]

Although the invention has been described above by reference to the graphically presented embodiment, the invention is not limited to the embodiment described above. For example, the present invention can be applied in an instance where the first planetary gearset G1 is a double-pinion planetary gearset with two sets of planet-pinion gears where the rotation input member is the first carrier PC1, and the rotation output member is the first ring gear R1.

[0056]

Although the embodiment has been described such that the transmission for an automatic transmission is constructed by the three planetary gearsets of one reduction planetary gearset and two planetary gearsets construction the compound planetary gear train, the transmission for an automatic transmission according to the present invention is not limited to the three planetary gearsets and can be applied to an automatic transmission where there are more than three planetary gearsets.

[Brief Explanation of Drawings]

[Fig. 1] It is a skeleton diagram schematically showing a gear transmission for an automatic transmission of an embodiment of the present invention.

[Fig. 2] It is a engagement logic explanatory view showing a relationship between selectable gears and engagement of transmission friction elements of the gear transmission.

[Fig. 3] It shows a torque transmission path of each gear of the gear transmission, wherein (a) is a skeleton diagram showing a torque flow path in a first forward gear, which is similar to Fig. 1, (b) is a skeleton diagram showing a torque flow path in a second forward gear, which is similar to Fig. 1, and (c) is a skeleton diagram showing a torque flow path in a third forward gear, which is similar to Fig. 1.

[Fig. 4] It shows a torque transmission path of each gear of the gear transmission, wherein (a) is a skeleton diagram showing a torque flow path in a fourth forward gear, which is similar to Fig. 1, (b) is a skeleton diagram showing a torque flow path in a fifth forward gear, which is similar to Fig. 1, and (c) is a skeleton diagram showing a torque flow path in a sixth forward gear, which is similar to Fig. 1.

[Fig. 5] It is a skeleton diagram showing a torque flow path in a reverse gear, which is similar to Fig. 1.

[Fig. 6] It is an exploded cross-sectional view showing the construction of the gear transmission shown in Figs. 1 through 5.

[Fig. 7] It is a skeleton diagram schematically showing physical location of members of the gear transmission.

[Fig. 8] It is an enlarged cross-sectional diagram showing detail of a part relating the present invention in the embodiment of the gear transmission.

[Fig. 9] It is an explanatory view for explaining a conventional construction of a gear transmission using a Ravigneaux compound planetary gear train.

[Explanation of Marks]

G1 first planetary gearset (planetary gear set for speed reduction)

G2 second planetary gearset (reverse gear shifting mechanism)

G3 third planetary gearset (reverse gear shifting9

M1 first connecting member

M2 second connecting member

C1 first clutch

C2 second clutch

C3 third clutch (direct clutch)

B1 first brake

B2 second brake

Input input member

1 input shaft

Output output member

2 output shaft

S1 first sun gear

R1 first ring gear (rotation input member: rotation output member)

P1 first pinion

PC1 first carrier PC1 (rotation output member: rotation input member

S2 second sun gear

R2 second ring gear

P2 second pinion

PC2 second carrier

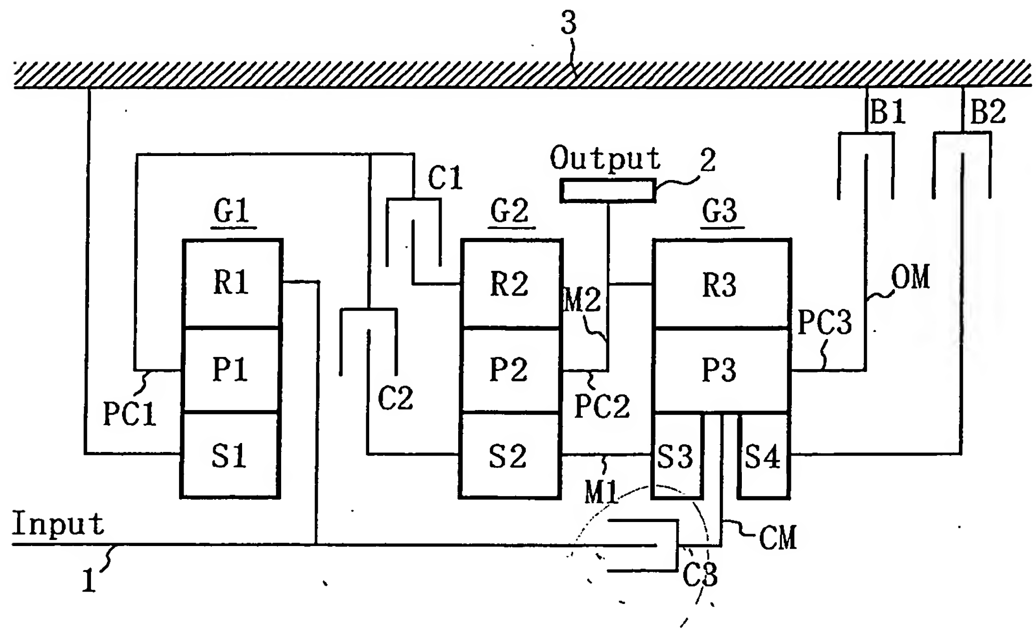
S3 third sun gear

S4 third pinion
P3 third pinion
PC3 third carrier
R3 third ring gear
CM center member
OM outer member
ENG engine (power source)
T/C torque converter
3 transmission case
4 middle shaft
5 pump housing (oil pump case)
6 pump cover (oil pump case)
6a sun gear fixing center boss portion
7 end cover
8 midway wall (output gear supporting wall)
9 hollow shaft
10 flange
11 clutch drum
12 clutch plate
13 clutch piston
14 third cutch fluid passage
15 clutch drum
16 clutch plate
17 clutch hub
18 clutch plate
19 clutch piston
20 clutch piston
21 first clutch or second clutch fluid passage
22 cylindrical connecting member (output drum)
23 brake hub
24 brake plate

25	brake piston
26	brake hub
27	brake plate
28	brake piston
29	counter shaft
30	counter gear
31	final drive pinion
32	clutch hub
51	opening member
52	opening member
52	connecting member

【書類名】　図面

【図１】 [Fig. 1]



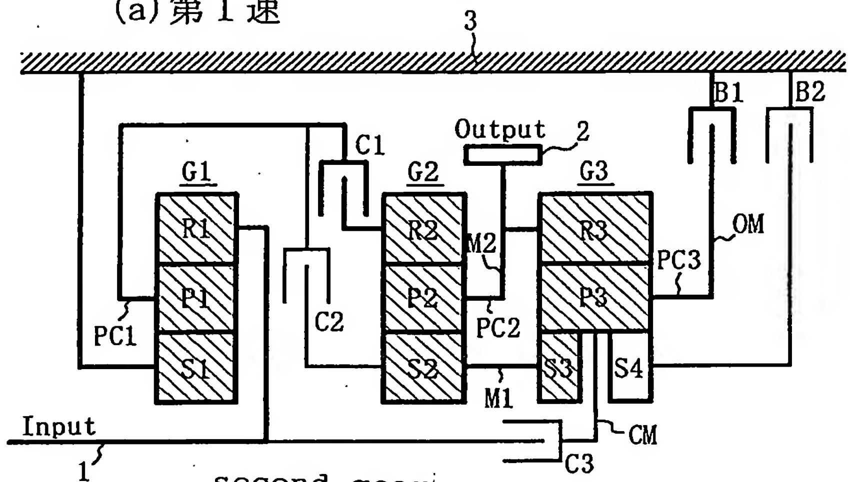
【図２】 [Fig. 2]

friction element

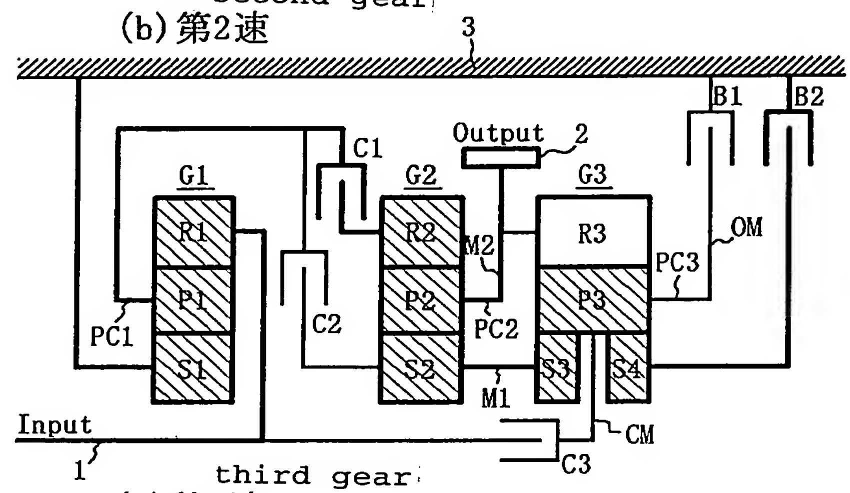
変速段 (gear) ＼ 摩擦要素		C1	C2	C3	B1	B2	
前進 (forward)	第1速	○			○		first gear
	第2速	○				○	second gear
	第3速	○	○				third gear
	第4速	○		○			fourth gear
	第5速		○	○			fifth gear
	第6速			○		○	sixth gear
後退			○		○		reverse gear

【図３】 [Fig. 3]

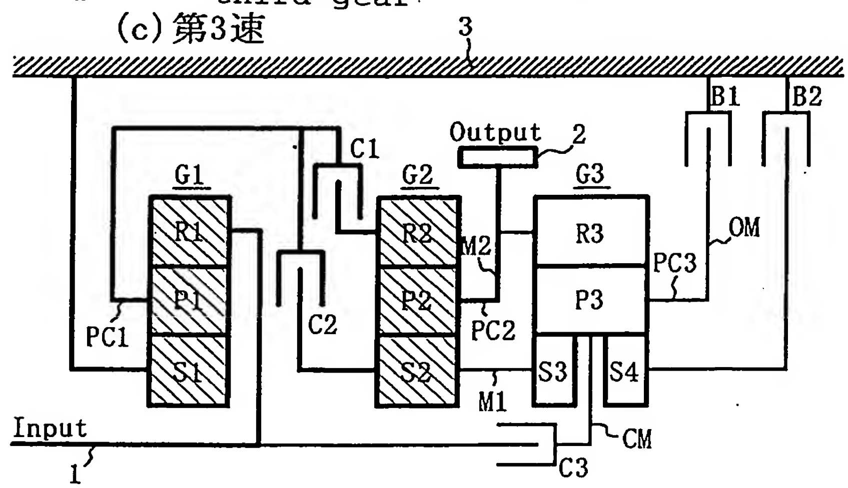
(a) 第１速 first gear



(b) 第2速 second gear

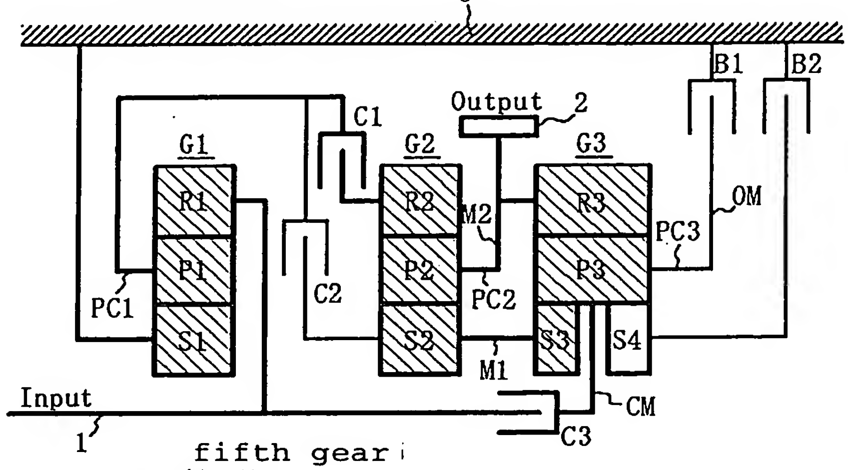


(c) 第3速 third gear

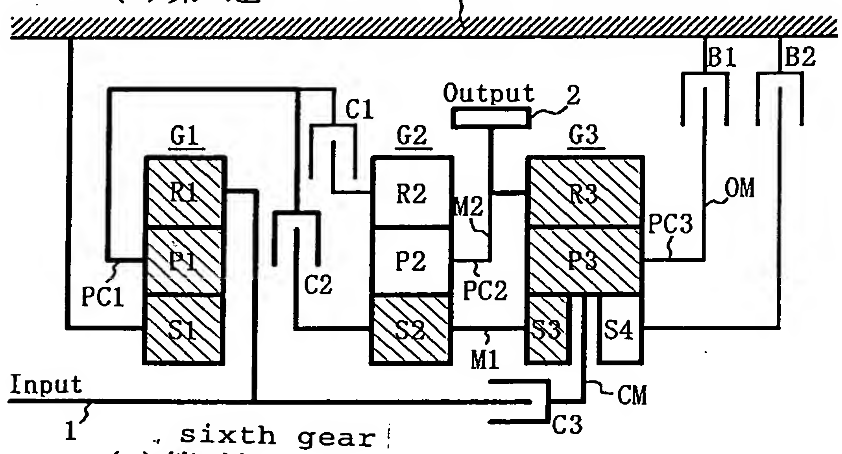


【図４】 [Fig. 4]

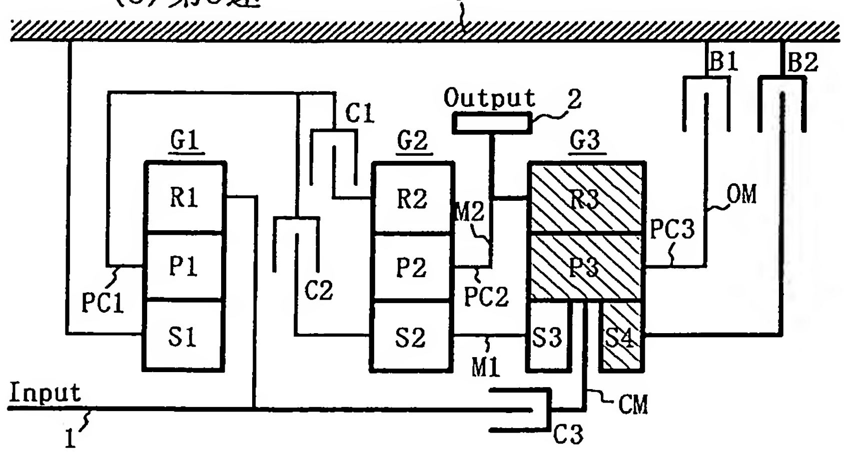
fourth gear
(a) 第4速



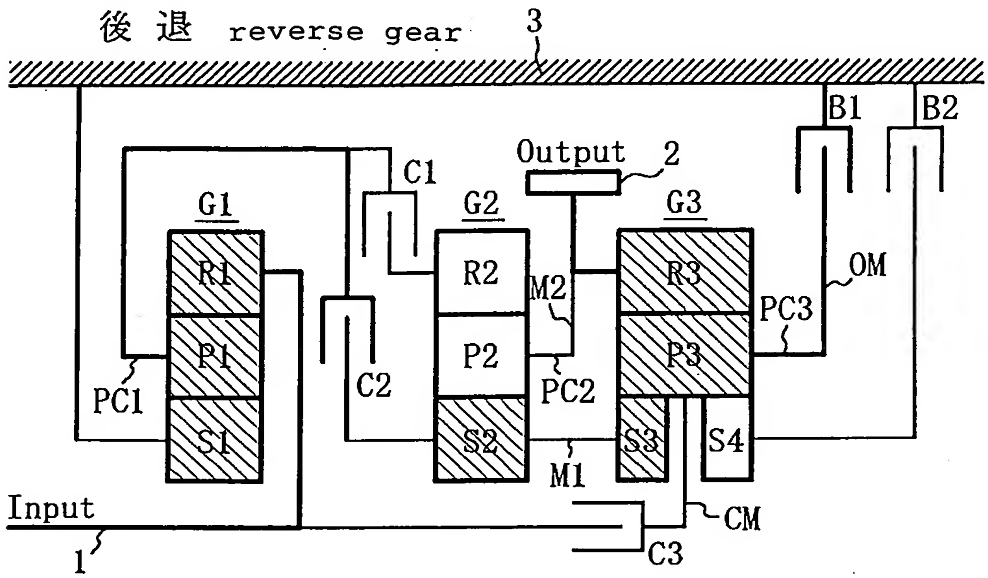
fifth gear
(b) 第5速



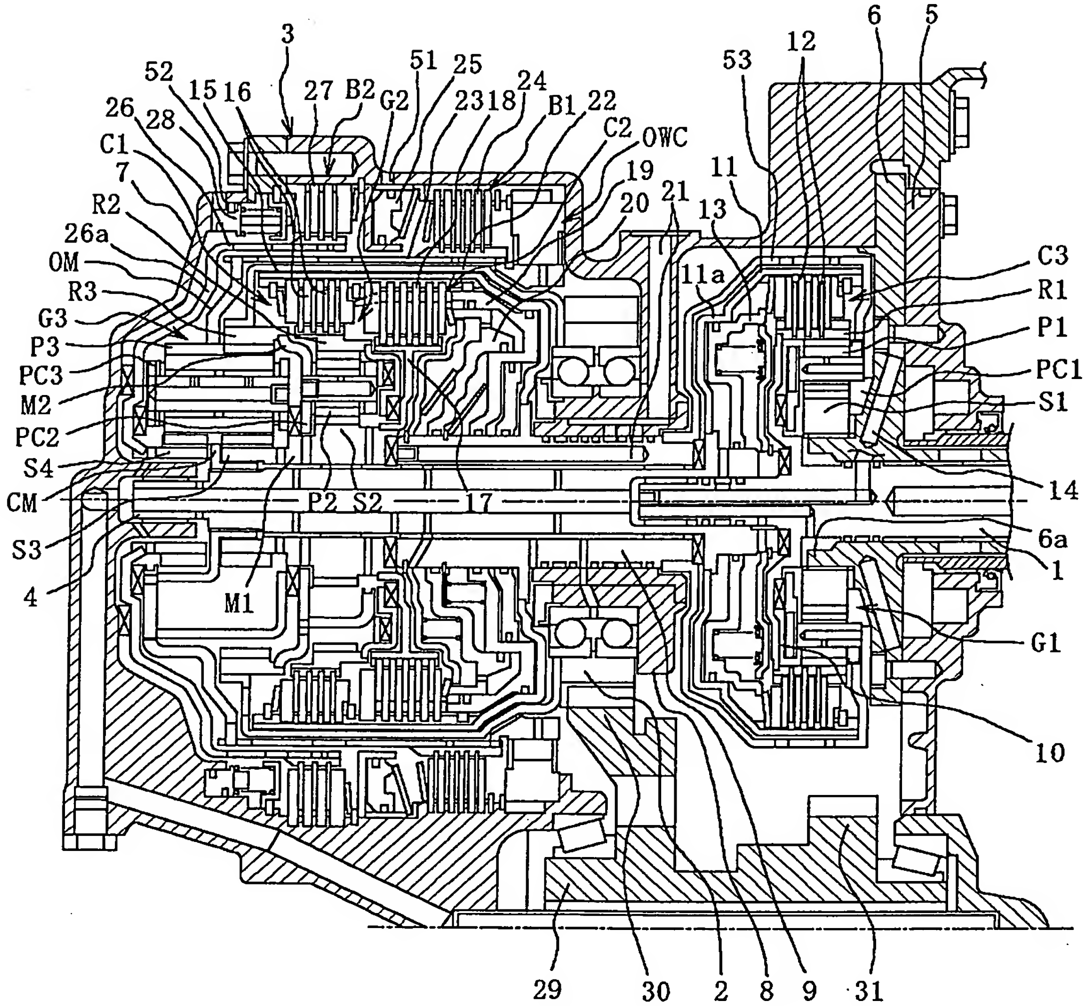
sixth gear
(c) 第6速



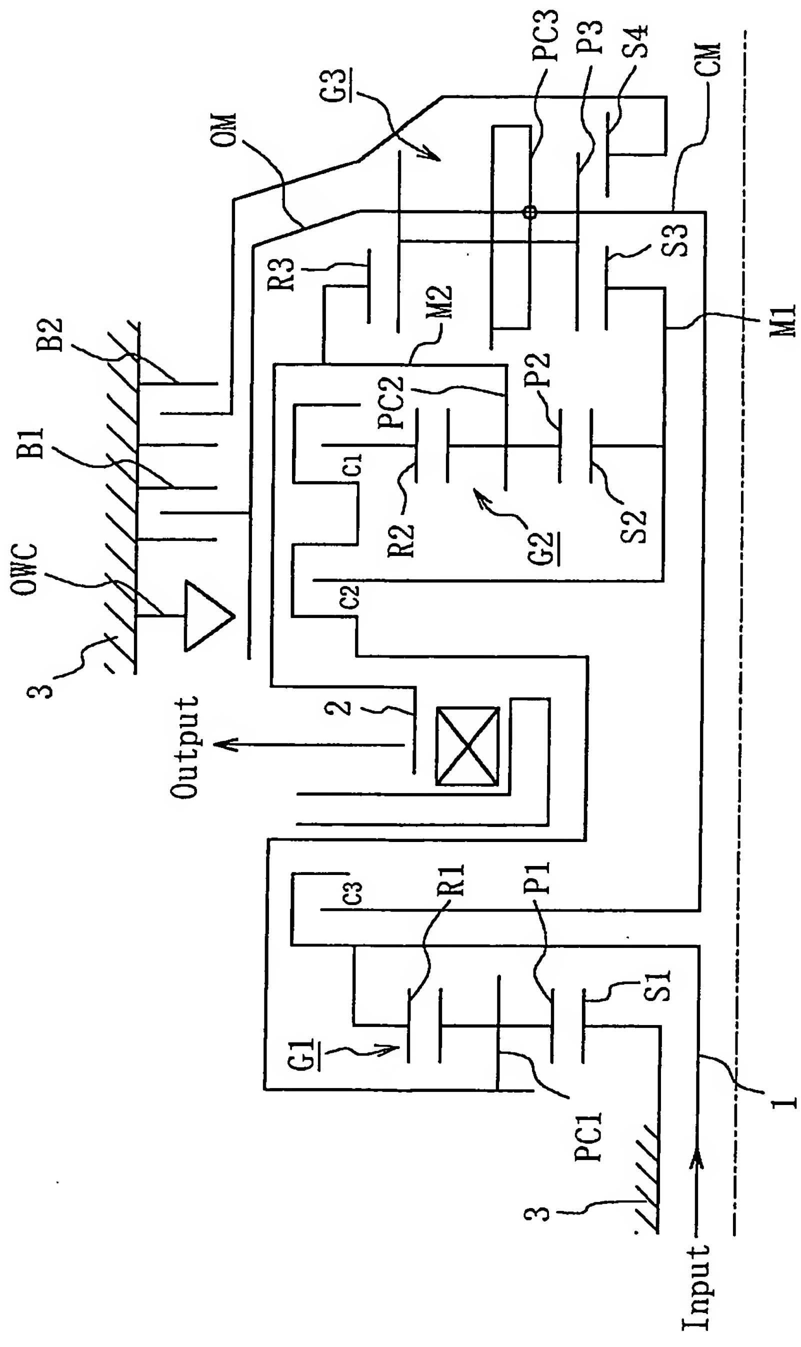
【図５】　[Fig. 5]


後 退 reverse gear
3
B1
B2
Output
2
C1
G1
G2
G3
R1
R2
R3
M2
OM
PC3
P1
P2
P3
PC1
C2
PC2
S1
S2
S3
S4
M1
Input
CM
1
C3

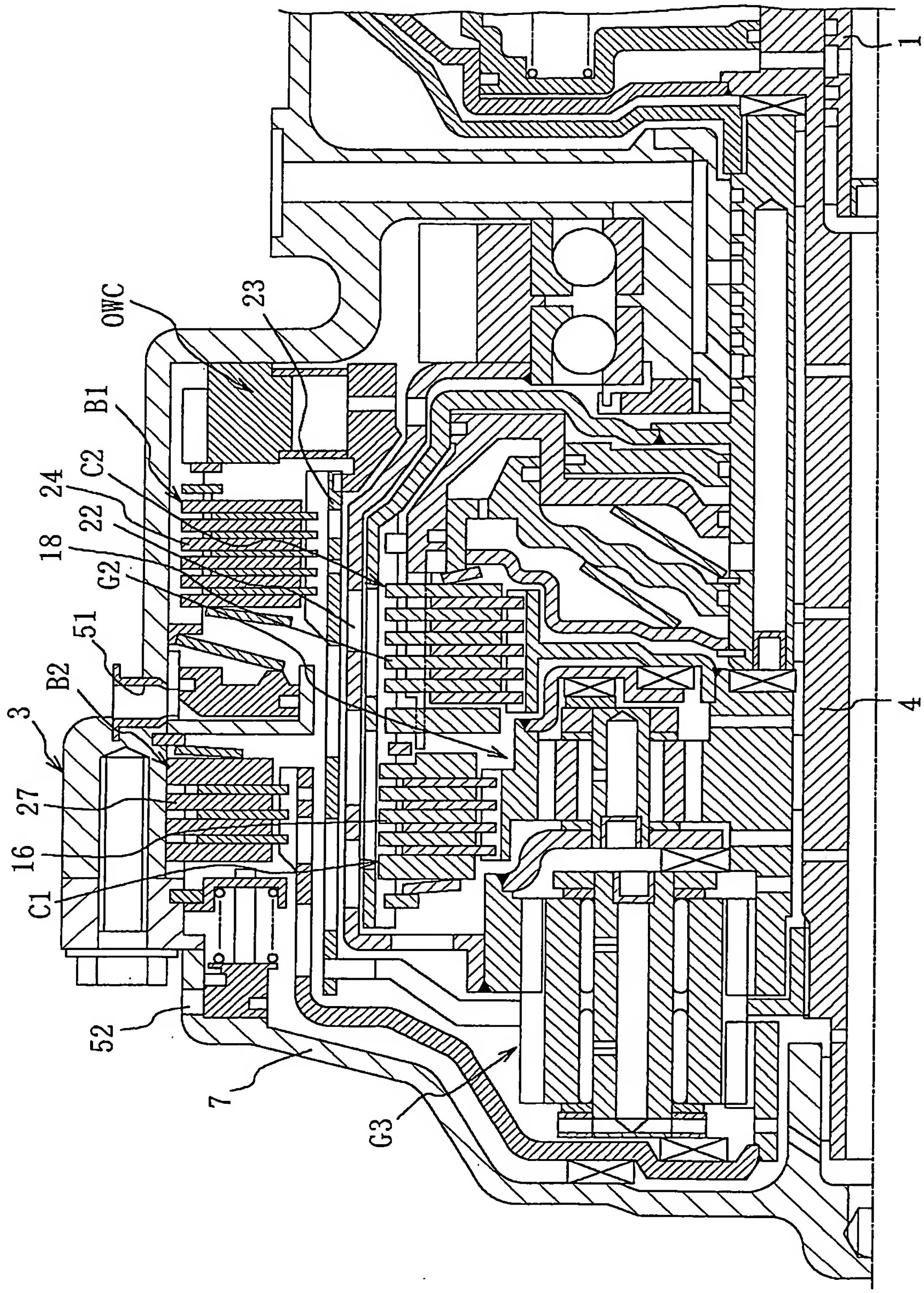
【図６】


3
52
15 16
27 B2
G2
51 25
23 18
24
B1 22
C2
OWC
26
28
C1
7
R2
26a
OM
R3
G3
P3
PC3
M2
PC2
S4
CM
S3
4
M1
P2 S2
17
19
20
21
11
13
11a
53
12
6
5
C3
R1
P1
PC1
S1
14
6a
1
G1
10
29
30
2
8
9
31

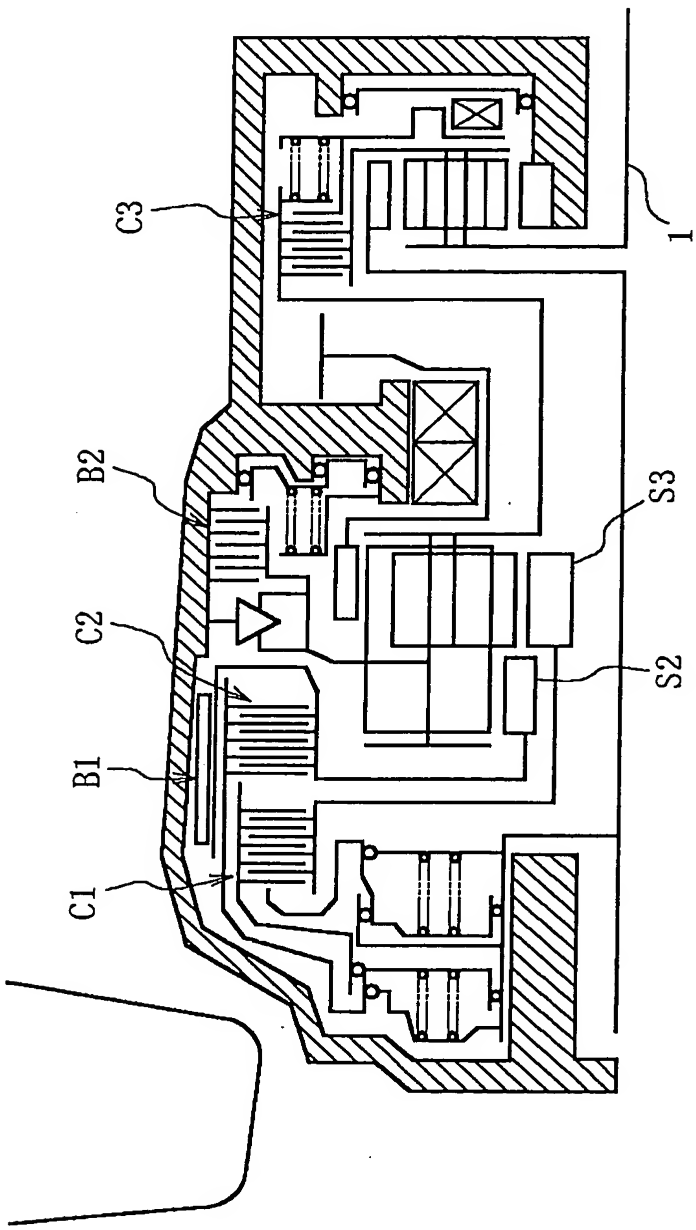
【図７】


G3
OM
PC3
P3
S4
CM
R3
M2
S3
M1
B2
B1
PC2
P2
C1
R2
G2
S2
OWC
C2
3
2
Output
C3
R1
P1
S1
G1
PC1
1
3
Input

【図８】



【図９】



[Document Name] Summary

[Summary]

[Problem] It is an object to improve compactness of a gear transmission which comprises a compound planetary train and is capable of select at least six forward gears and on reverse gear.

[Solving Means] A gear transmission comprises a reduction planetary gearset G1, a single-pinion type planetary gearset G1 and a double-sun-gear type planetary gearset G3 which are disposed in row in the order of mention. The gear transmission is capable of realize six forward gears including O/D gear by these gearsets and clutches C1 through C3 and brakes B1 and B2 while an input shaft 1 and an output gear 2 are coaxially arranged. The gear transmission is arranged such that a ring gear R2 serves as an input member of reduced rotation from the reduction planetary gearset G1 and that the brakes B1 and B3 are disposed around the clutches C1 and C2 so that the clutch C1 and the brake B2 overlap at least partially in the axial direction and the clutch C2 and the brake B1 overlap at least partially in the axial direction.

[Selected Figure] Fig. 6.